MEASURED EFFECTS OF RETROFITS - A REFRIGERANT OIL ADDITIVE AND A CONDENSER SPRAY DEVICE - ON THE COOLING PERFORMANCE OF A HEAT PUMP.

William P. Levins, James R. Sand, Van D. Baxter, Randall S. Linkous
Staff Members
Oak Ridge National Laboratory, Oak Ridge, TN

ABSTRACT

A 15-year old, 3-ton single package air-to-air heat pump was tested in laboratory environmental chambers simulating indoor and outdoor conditions. After documenting initial performance, the unit was retrofitted with a prototype condenser water-spray device and retested. Results at standard ARI cooling rating conditions (95°F outdoor dry bulb and 80/67°F indoor dry bulb/wet bulb temperatures) showed the capacity increased by about 7%, and the electric power demand dropped by about 8%, resulting in a steady-state EER increase of 17%. Suction and discharge pressures were reduced by 7 and 37 psi, respectively.

A refrigerant oil additive formulated to enhance refrigerant-side heat transfer was added at a dose of one ounce per ton of rated capacity, and the unit was tested for several days at the same 95°F outdoor conditions and showed essentially no increase in capacity, and a slight 3% increase in steady-state EER. Adding more additive lowered the EER slightly. Suction and discharge pressures were essentially unchanged.

Our short-term testing showed that the condenser-spray device was effective in increasing the cooling capacity and lowering the electrical demand on an old and relatively inefficient heat pump, but the refrigerant additive had little effect on the cooling performance of our unit. Sprayer issues to be resolved include the effect of a sprayer on a new, high-efficiency air conditioner/heat pump, reliable long-term operation, and economics.

INTRODUCTION

A number of studies have been undertaken to investigate the effects of retrofitting building thermal envelopes such as attic, wall, and foundation insulation/upgrades, air-leakage control, furnace and air conditioner replacements and tune-ups, improved windows and doors, etc.(Levins 1994; Ternes 1991,1992; Brown 1993; Parker 1993, 1994). However, relatively little attention has been paid to the potential impact that retrofits of existing air-conditioning/heating equipment could have on building energy usage. The Department of Energy's Existing Buildings Efficiency Research (EBER) Program sponsored a study in 1995 to evaluate the impact of replacing the existing compressor in a residential air-conditioner with a smaller, high-efficiency model (Levins 1996).

A used, 15-year old, nominal 3-ton, single-package heat pump was purchased from a local heating and air-conditioning contractor. The unit was cleaned up, properly charged, and tested in the cooling mode in order to establish the heat pump's baseline "as received" cooling performance. Testing was done in a set of environmental chambers at standard rating conditions as specified in Air-Conditioning and Refrigeration Institute (ARI) Standard 210/240-89 (ARI, 1989). A state-of-the-art reciprocating compressor with 30% less capacity (the system cooling capacity was reduced by 20%) was then installed in the old heat pump and the unit was retested. Modeling of the unit predicted an increase in EER of about 30%. A 33% increase in EER was achieved in the tests along with a 38% decrease in electric power demand.

Subsequently, the original compressor was reinstalled and two somewhat simpler retrofit measures were tested. The first was a stand-alone device designed to spray a fine mist of water on the outdoor (condenser) coil. The spray coil was removed and a proprietary refrigeration oil additive, formulated to enhance the refrigerant-side heat transfer performance as well as reduce friction in the compressor, was added to the heat pump oil charge and tested. This paper discusses the results of tests performed with these two retrofit measures.

CONDENSER SPRAY UNIT

Background

A condenser spray system sprays water on the air-side of a refrigerant-to-air heat exchanger and increases its efficiency by lowering the condensing temperature (and pressure) of the refrigerant inside the tubes. This action lowers the pressure ratio across a compressor, thereby reducing its power demand. Spray systems should be particularly effective on hot days and on inefficient/undersized heat exchangers.

The condenser spray unit we tested in our environmental chambers was a prototype design being developed by a small manufacturer in the Southeast U. S. It consists of a control unit, tubing, and one or more spray nozzles. The control unit is approximately 6" x 6" x 4" high in outer dimensions. It is usually connected to the building water supply and controls water delivery to the coil face with a solenoid valve that requires a 24 volt AC supply.

Power to the sprayer can be supplied from the 24 volt control circuit of the heat pump, or it can be obtained from an optional photovoltaic (PV) cell mounted in the unit cover and coupled to a battery backup. The PV cell/battery approach is quite attractive from the standpoint of convenient installation. The manufacturer is working on design issues concerning reliable operation of the battery and the amount of sunlight needed for reliable operation. We supplied power to the sprayer unit from a variable-voltage transformer set to deliver 24 volts AC for our laboratory tests. The manufacturer said the control unit was equipped with a sensor to detect vibration of the heat pump outdoor fan so that the water solenoid valve would only be energized when the heat pump was operating. The vibration sensor in the particular unit we tested did not function properly. However, since we only ran steady-state tests, it was not a problem for us.

Figures 1 and 2 show an evaluation unit in field operation on an air conditioner in a building near our laboratory. This installation used four nozzles and was part of a field trial conducted by another group after we had conducted our laboratory tests. They had problems with the PV cell/battery power supply and had to connect the sprayer to the control power of the air-conditioner for proper operation. Their test results were similar to those we achieved (Kevil, 1995, personal communication to V. Baxter). They observed some problem with plugging of the spray nozzles during their testing and also noted some scaling on the condenser coil surface after the tests. Some revised nozzle designs are being investigated by the manufacturer to address the plugging problem.

Test Setup

Figure 3 is a photograph of the 15 year-old heat pump used for testing after it was externally washed and the heat exchangers were cleaned. The heat pump was instrumented with thermocouples, thermopiles, pressure transducers, and watt transducers at all points of interest for monitoring. Refrigerant temperatures were measured with type-T thermocouples strapped to tube walls and covered with insulating tape. Sensible capacity was calculated from air-side temperature measurements using 9-point type-T thermopiles and an airflow measurement from a parallel-cell, honeycombed grid with a multi-point pitot tube array, located on the inlet air side. Latent capacity was measured by collecting and weighing condensate. The monitoring system scanned each channel every five seconds and output one-minute averaged readings to a data file. Pressure transducers and the airflow measuring array were calibrated by our instrument technicians and the heat pump and the climate chambers were checked to be sure there was no air leakage between indoor and outdoor sections.

Figure 4 is a schematic diagram showing the condenser sprayer as installed on the test heat pump, along with thermocouple locations used for performance monitoring. All testing was conducted in a set of temperature and humidity controllable environmental chambers located in our laboratory.

The condenser spray unit was installed by the manufacturer's representatives on our heat pump. We attached a hose from a standard outdoor faucet to the spray unit using fittings supplied by the manufacturer. We used both one spray nozzle and two spray nozzles in our tests. The unit can supply water to more than two nozzles provided there is sufficient water pressure. We investigated several spray patterns to determine the effect of this parameter on performance, but only the number of nozzles had any measurable impact. The nozzle spray cone was about seven inches in diameter when the nozzle was about six inches from the condenser surface -- Figure 5 shows one of the plastic nozzles, and Figure 6 shows the spray pattern on a test piece of blotting paper. Total water consumption of the sprayer we tested was about 0.02 gpm per nozzle at 55 psig line pressure-- about 3600 gallons of water for a location with a long cooling season of 3000 operating hours (assuming full-time operation of the sprayer).

Results

We characterized the cooling mode performance of the heat pump at standard ARI rating conditions of 95°F outdoor temperature and 80/67°F indoor dry/wet bulb temperatures, with no water spray from the nozzles. Capacity, EER, and suction and discharge pressures were essentially unchanged from previous testing we had done with the unit -- 31,600 Btu/h, 5.74 Btu/watt, 73 psig, and 303 psig, respectively. We activated two nozzles and operated the unit continuously for four hours, continuing our testing on the following day with one nozzle activated.

In our haste to get the sprayer installed and operational, we failed to measure the condensate coming from the evaporator coil during this (condenser-spray device) testing in order to obtain the latent load. However, we had a great deal of data from testing done in the two preceding months (for another project) without the sprayer, so we were able to estimate the latent load at that condition with an excellent degree of confidence. However, since the evaporator pressure (and hence its temperature) dropped when the sprayer was operational, we suspected that the sensible to total (S/T) load ratio might be lower for these cases. The measured evaporator and condenser pressures were 70/281 psig, respectively, with a single nozzle operating, and 66/266 psig, respectively, with two nozzles operating.

We had modeled this particular heat pump in the cooling mode in previous work (Levins 1996), and used measured data to calibrate the model. We used the same heat pump design model (Fisher and Rice, 1991) along with measured pressures, temperatures, etc. from these latest runs, and estimated that the S/T for one nozzle should be 0.716 compared to a measured no nozzles S/T of 0.726. With two nozzles operating the S/T was estimated to be 0.691. Based on the measured sensible loads and the estimated latent loads for the system with the nozzles operating, we predicted a total capacity increase over our original air conditioner of about 1.0% with one nozzle operating, and about 7.5% with two nozzles operating. Table 1 contains the results of our testing with the predicted latent capacities for the two cases when the sprayer was operating.

The steady-state EER increased by about 6% with one nozzle spraying water on the condenser and by about 17% when two nozzles were operating. The capacity and efficiency gains are substantial for the system when two nozzles are operating -- most probably because the increased condenser area coverage was needed for this particular system.

Conclusions

Our short-term testing showed that a condenser spray device appears to be effective when mounted on an old, relatively inefficient heat pump operating in the cooling mode. The sprayer may not be as effective when mounted on a new high-efficiency air conditioner -- this is an area that should be researched. The economics of such a device would appear to be more favorable in a location with a high number of cooling hours. Utilities would benefit from the electric power demand reduction that the sprayers would provide on peak days.

Other areas that warrant more study are the formation of scale on the fins and tubing of the condenser from dissolved solids in the water, the possible plugging of the sprayer nozzle itself, the performance of the sprayer under different outdoor temperatures and humidities, the effect of pollen and dust on the fouling of a sprayed condenser, seasonal testing to estimate a seasonal EER (SEER) in several climates, and a method of disconnecting/draining the sprayer in winter weather to avoid freeze-up.

Manufacturers of condenser sprayers must provide reliable controllers that are able to adapt to each air conditioner environment in the most efficient manner.

Table 1. Effect of Condenser Water Spray on Cooling Performance of Heat Pump @ 95°F

Number of Nozzles	Total Electric Power (Watts)	Capacity (kBtu/h)			Steady-State EER (Btu/Watt)	cfm/Ton	Water Spray Rate (gpm)
		Sensible	Latent*	Total			
0	5513	23.0	8.7	31.6	5.74	429	0
1	5248	22.9	9.1	32.0	6.10	424	0.02
2	5053	23.5	10.5	34.0	6.73	401	0.04

*Note: Latent capacities were estimated from modeling using operating parameters for the 1 and 2 nozzle tests.

REFRIGERANT OIL ADDITIVE

Background

Refrigerant oil chemical additives are often-advertised products that claim to improve the performance of older heat pumps and air conditioning equipment by strongly adsorbing on inner metal surfaces of a system via chemisorption. Similar longer chain, polar molecules with an electron rich functional group on one end and an electron deficient moiety on the other end had previously been used as boundary layer or extreme pressure (EP) additives in lubricating oils. They provide a tightly adhering monolayer of organic material on metal surfaces, and help avoid metal-to-metal surface contacts associated with mechanical equipment start-up and shut-down. These additives supposedly reduce compressor wear and save energy by reducing mechanical friction.

Since these compounds also tenaciously adsorb to the inner walls of heat exchanger tubing, they reportedly function as a heat transfer enhancing additive -- they increase the refrigerant-side heat transfer coefficient, thereby improving the overall heat exchanger efficiency. Advertising literature states that chemical heat transfer additives function by forming a monolayer on the inside walls of heat exchanger tubing, displacing a build-up of oil that acts as a barrier to efficient heat transfer.

Such chemical additives have strong popular appeal because their cost is minimal when compared to other hardware options, and their use requires little or no system hardware modifications.

Experimental Setup

A nationally advertised chemical heat transfer additive based on a chlorinated α-olefin active ingredient compounded with a chlorine stabilizer (to prevent free chloride ion formation and system corrosion) and dissolved in a mineral oil carrier solution (U.S. Patent # 4,963,280, October 16, 1990) was tested at the ARI 95°F air conditioning rating point in the 15-year old heat pump as described previously. After reproducible operational data and a coefficient of performance (COP) and energy efficiency ratio (EER) based on sensible and latent performance were established at this rating condition, 3.0 ounces (the manufacturers recommended amount) of the refrigerant oil additive were added to the system.

The system was allowed to operate with cycling overnight to allow additive distribution throughout the system. The heat pump was then operated at steady-state conditions over a six-day period, and its performance was monitored several times a day, for 15 minute intervals.

We added another 1.5 ounces to the system seven days after the initial additive addition, at the recommendation of the additive supplier. System performance monitoring continued as before. On the last day of testing, day 9, we added another ounce of additive in the morning and the unit was run continuously during the day, with steady-state performance data again collected as before.

Results

Table 2 contains the results of our steady-state testing at the ARI 95°F outdoor air rating point. The suction pressures varied from 73 to 70 psig during the testing, while the discharge pressures were essentially constant for all runs at 300 psig. The indoor airflow rate was held constant for all runs at about 1120 cfm.

Table 2 indicates that no improvement in heat pump performance was measured in our laboratory tests as a result of adding this product to our test unit. The small changes (± 2%) in steady state compressor power consumption and cooling capacity shown in Table 2 are most likely attributable to random experimental errors, although a small 2.5% improvement in EER is indicated for 3 ounces of the additive. It is also worth noting that the evaporator air entering/leaving temperatures and the compressor pressure ratio showed no significant change as a result of additive addition. Both of these observations are consistent with no improvement in heat exchanger performance. There was, however, a noticeable, but unquantified, decrease of compressor noise resulting from additive addition.

Analysis and Conclusions

An estimate of the relative energy losses due to the heat exchanger inefficiencies in a typical air-to-air residential heat pump is about 28%; with about 18% of this loss in the evaporater and 10% in the condenser (Bullock 1988) (see Figure 7). A further breakdown of the thermal resistances contributing to heat exchanger inefficiency is shown schematically in Figure 8. It shows that for a flat plate fin heat exchanger with smooth internal tube walls, roughly half of the overall (air-to-metal-to-refrigerant) thermal resistance of the coil is due to air-side resistance, whereas the refrigerant-side resistance contributes only about a third of the total. Hence, a 50% decrease in the refrigerant-side heat transfer resistance would be needed to effect a 17% improvement in overall coil performance. Only about a 30% improvement in the air-side heat transfer resistance would be needed to accomplish the same net gain in coil performance (Bullock 1988). The heat exchangers are air-side controlled for such systems.

Table 2. Effect of Refrigerant Oil Additive on Cooling Performance of Heat Pump @ 95°F

Ounces of Additive	Total Electric Power (Watts)	Capacity (kBtu/h)			Steady-State EER (Btu/Watt)	cfm/Ton	Compressor Pressure Ratio
		Sensible	Latent	Total			
0	5485	23.2	8.7	31.9	5.82	422	4.12
3.0	5428	23.7	8.7	32.4	5.97	416	4.15
4.5	5403	23.3	8.4	31.6	5.86	423	4.29
5.5	5385	23.1	8.3	31.5	5.84	428	4.27

Note: Latent capacities were measured by collecting condensate from evaporator.

It was disappointing, but not surprising, that no efficiency improvements were measured as a result of theses additive tests. Since only 28% of the total heat pump energy use can be attributed to inefficiencies in heat exchanger performance, and the dominant factor in that inefficiency is air-side heat transfer, it would take a remarkable improvement in the refrigerant-side heat transfer (80 to 100%) to improve the overall, steady-state, heat pump performance by 8 to 10%. Clearly a more refined testing procedure focusing on the value of the refrigerant-side heat transfer coefficient is needed to accurately evaluate additive performance. System efficiency/performance improvements may be more evident in applications of this product to chillers or similar products with water-to-refrigerant heat exchangers, where there is a better correspondence between water-side and refrigerant-side thermal resistances.

Another claim that can be quantitatively analyzed is that these strongly adsorbed molecules function as "mini-fins" to increase refrigerant-side heat transfer efficiency. General formulas for calculating overall heat transfer coefficients and heat exchanger fin effectiveness for thermal circuits can be found in Kays and London, *Compact Heat Exchangers*. Using formulas for fin effectiveness and temperature effectiveness of heat transfer surfaces with dimensions appropriate for a molecular spine fin, a fin effectiveness, η_f, very close to unity (1.0) is calculated. This results in a surface temperature effectiveness essentially identical to the smooth tube surface. If a true monolayer of adsorbed additive is present on the surface of these tubes, it could be considered as a very minor fouling of the surface -- an additional layer through which heat must be conducted. If this monolayer prevents the formation of other foulants or less conductive surface films, however, it will aid the overall heat transfer process.

Based on the previous discussions, we conclude that we should not have measured any significant changes in our air-to air heat pump performance as the result of adding a refrigerant-side heat-transfer enhancer to our system. Our testing measurements uphold our after-the-fact analysis of the situation. The additive should, however, be tested in a refrigeration system with water-to-refrigerant heat exchanger(s) to verify its performance there.

We also measured no significant decrease in compressor power due to reduced mechanical friction with the additive, even though the compressor appeared to operate with less noise. The compressor pressure ratio increased only slightly as a result of the presence of the additive. Therefore, we conclude that mechanical friction is not reduced by the additive in our system.

ACKNOWLEDGMENTS

Appreciation is expressed to the Department of Energy, Office of Building Technologies, for sponsoring the bulk of this work under contract #DE-AC05-96OR22464 with Lockheed-Martin Energy Systems.

The authors also thank Tom Kevil of the Oak Ridge Centers for Manufacturing Technology for their sponsorship of the tests of the condenser spray unit.

We also express our thanks to Keith Rice of Oak Ridge National Laboratories for the modeling work he performed to estimate the latent capacities used in the condenser sprayer testing.

REFERENCES

1. Air-Conditioning and Refrigeration Institute, "*Standard For Unitary Air-Conditioning And Air-Source Heat Pump Equipment*", Standard 210/240-89, 1989.

2. Brown, M. A. with L.J. Berry, R.A. Balzer, and E. Faby, "*National Impacts of the Weatherization Assistance Program in Single-Family and Small Multifamily Dwellings*", ORNL/CON-326, Oak Ridge National Laboratory, Oak Ridge, TN, 1993.

3. Bullock, Charles E., "*Recent Developments in Residential Heat Pump Technology*", Proceedings of the JAR International Symposium on Recent Developments in Heat Pump Technology, March 9-10, Tokyo, Japan, pp. 61-69, 1988.

4. Fischer, S. K. and C.K. Rice, "*The Oak Ridge Heat Pump Models: I. A Steady-State Computer Design Model For Air-To-Air Heat Pumps*", ORNL/CON-80/R1, August 1983.

5. Kays, W.M., and A.L. London, "*Compact Heat Exchangers*", 3rd edition, McGraw-Hill, New York, pp. 14-16.

6. Levins, W.P. with M.P. Ternes, "*Are Heating System Retrofit and Tune-up Programs Really Increasing the Efficiency of Oil-Fired Systems*", ACEEE Proceedings, Vol 5, pp143-151, August 28 - September 3, 1994

7. Levins,W.P.with C.K. Rice and V.D. Baxter, "*Modeled and Measured Effects of Compressor Downsizing in an Existing Air Conditioner/Heat Pump in the Cooling Mode*", ASHRAE Paper #H-1048, to be presented at the summer meeting, June, 1996.

8. Parker, D. with J. Cummings and A. Meier, "*Will Duct Repairs Reduce Cooling Loads*", Home Energy Magazine, Sept/Oct 1993.

9. Parker, D. with S. Barkazi, "*Saving Energy with Reflective Roof Coatings*", Home Energy Magazine, May/June 1994.

10. Ternes, M. P. with P.S. Hu, L.S. Williams, and P. Goewey, "*The National Fuel End-Use Efficiency Field Test: Energy Savings and Performance of an Improved Energy Conservation Measure Selection Technique*", ORNL/CON-303, Oak Ridge National Laboratory, Oak Ridge, TN, 1991.

11. Ternes, M.P. with W.P. Levins, "*The Oklahoma Field Test: Space Cooling Electricity Savings from Standard Energy Conservation Measures, Radiant Barriers, and High-Efficiency Window Air Conditioners*", ORNL/CON-317, 1992.

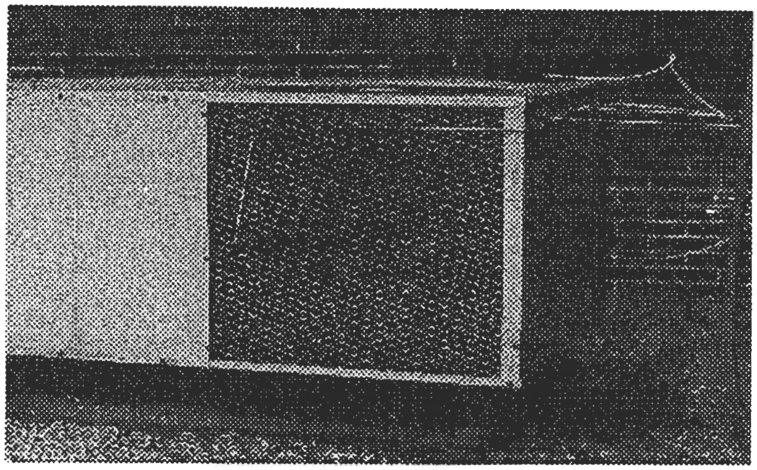

Figure 1 Field test evaluation of condenser spray unit.

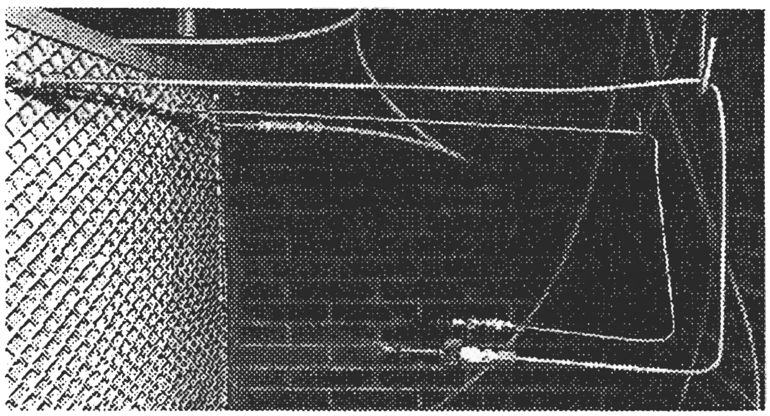

Figure 2 Close-up view of field test condenser spray unit in operation.

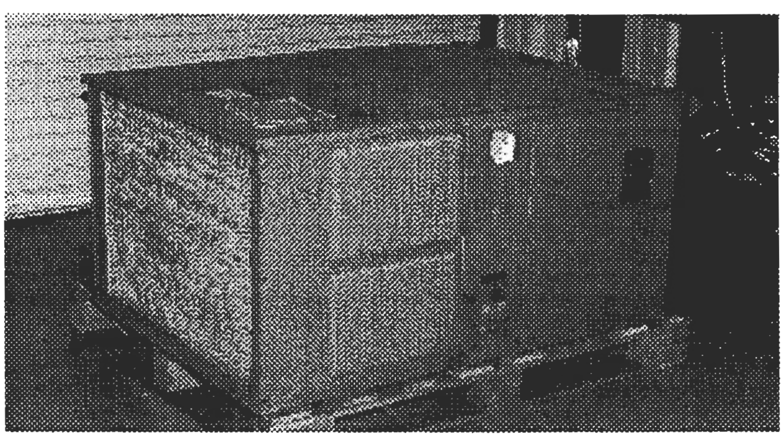

Figure 3 As-received single-package heat pump used in testing.

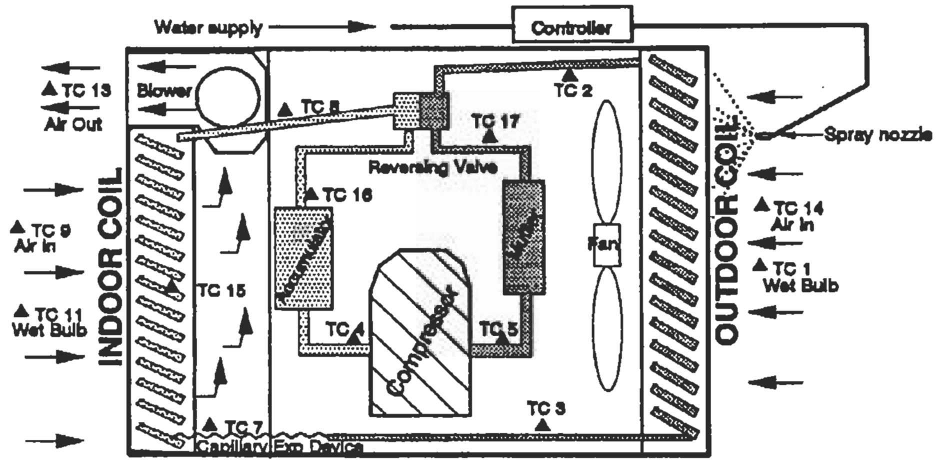

Figure 4 Schematic diagram of heat pump showing thermocouple locations.

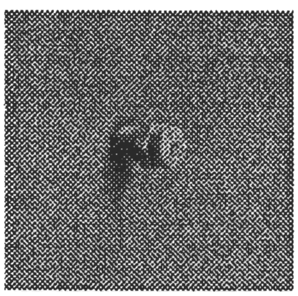

Figure 5 Plastic spray nozzle.

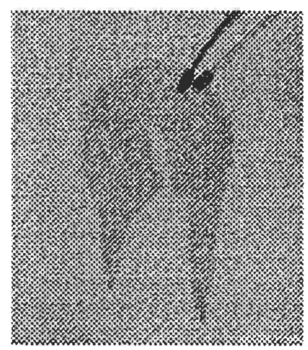

Figure 6 Spray pattern from nozzle on blotting paper.

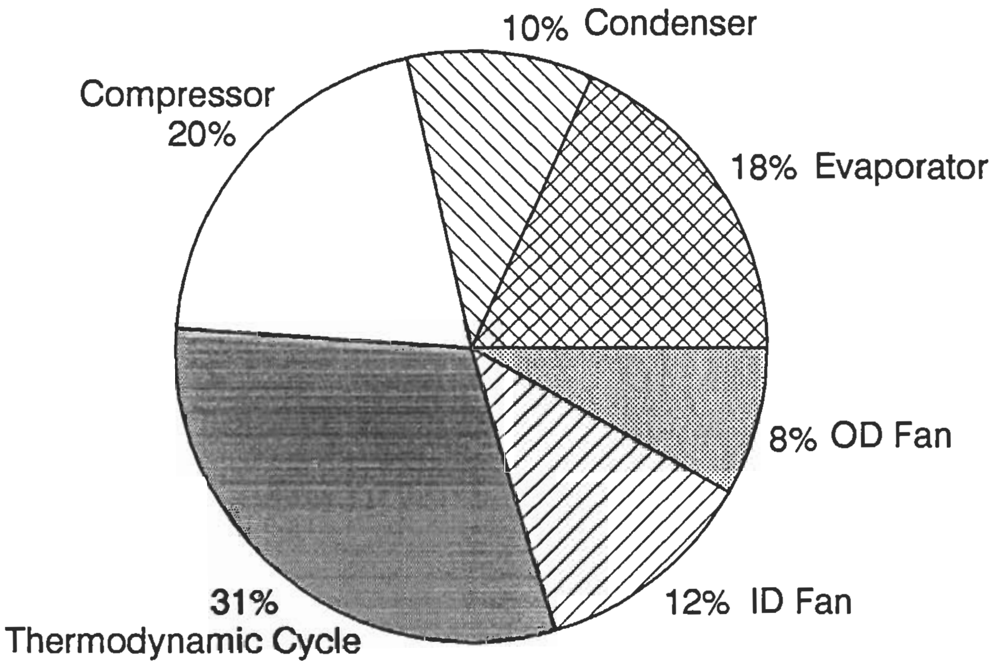

Figure 7 Energy use pattern of a typical air conditioner.

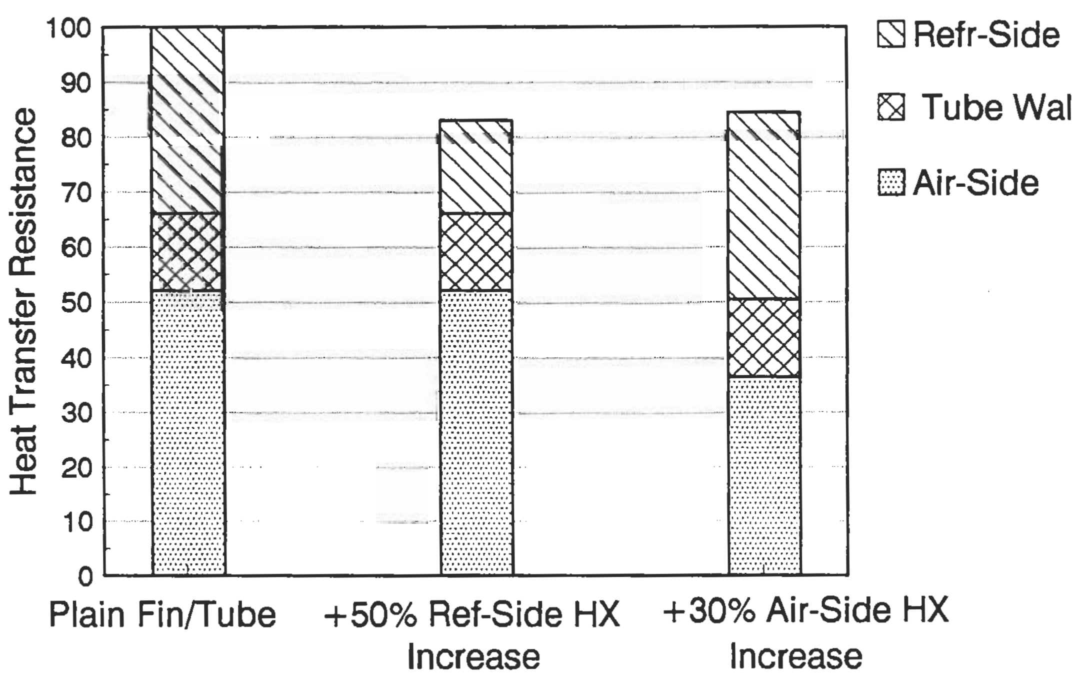

Figure 8 Component heat transfer resistances of a refrigerant-to-air heat exchanger.